Cryogenic Systems for the 3 TeV Injector Study

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I. CHOICE OF CONDUCTOR OPERATING TEMPERATURE

At the 1996 Snowmass workshop [10] several alternative cryogenic system arrangements were identified for different versions of the transmission-line magnet. For a NbTi conductor a system providing a conductor temperature between 4.5 and 5 K was suggested, and for NbSn the temperature range was 4.5 - 6.5 K. The first of these is of the SSC type with a transmission line flow of subcooled helium in series passing through recooler heat exchangers where heat is exchanged with boiling saturated helium at 4.5 K. In the second of these systems the transmission line is cooled by supercritical helium streams which are connected in parallel. There is no boiling helium in this second system. Instead heat is transported by the sensible heat of the supercritical stream, but operating just outside the critical region and expanding the stream as it passes through the transmission line produces a large effective heat capacity. Thus the temperature of the stream rises only 1.5 K while taking up heat of 28 J/g. The heat capacity of boiling helium in the saturated system is infinite and the temperature rise zero, but the latent heat is only 18 J/g.

The supercritical system requires lower flow rates for a given heat removal and tolerates larger pressure drops than the saturated system. In addition it is simpler, requiring no recoolers. The price that is paid for these advantages is a higher conductor operating temperature. Thus the trade-off is between current density in the superconductor and cost of cryogenics. In a high field magnet of small cross-section, high current density is an extremely important cost driver because the number of ampere-turns needed for a given central field increases as the current density decreases, and because the size of the coil drives the size of the whole magnet. In general, this trade-off in systems such as the Tevatron, SSC and LHC favors lower temperature and higher current density. In the Low-field transmission line magnet, however, all of the current density in magnet is equally effective in driving the magnet gaps, and the conductor current density does not drive the size of the magnet. The size of the conductor is instead dominated by the amount of copper needed to give adequate quench protection and the space needed for helium flow. The trade-off in this case, therefore, is strictly between the cost of the cryogenics and the cost of the additional NbTi in the conductor. This is a very different cost equation and it gives a different answer.

To make this concrete it is useful to use recent cost information from IGC. For 80,000 A at 4.2 K, this vendor suggests 23 strands each 1.30 mm in diameter with CuSc ratio 1.35:1. For this strand they quote $0.5844 per foot. It is not clear how much margin is included in this recommendation. Ignoring any included margin and assuming that we would like a critical current at 100,000 A, and taking a linear relationship of critical current with temperature, $I_c = I_o(1 - T/8.8)$, costs for conductor at various temperatures can be determined and are shown in Table I.

Table I
Magnet Transmission Line SC Cost and Quench Characteristics

For 100,000 A conductor, I_o = 6654 A for strand

Temp.	# strands	Cost per m	T for quench at 70,000 A
4.2 K	29	$63.95	5.60 K
5.0 K	35	$77.18	6.15 K
6.0 K	47	$103.64	6.83 K

A factor of 1.15 has been applied to these costs to provide for cable lay.

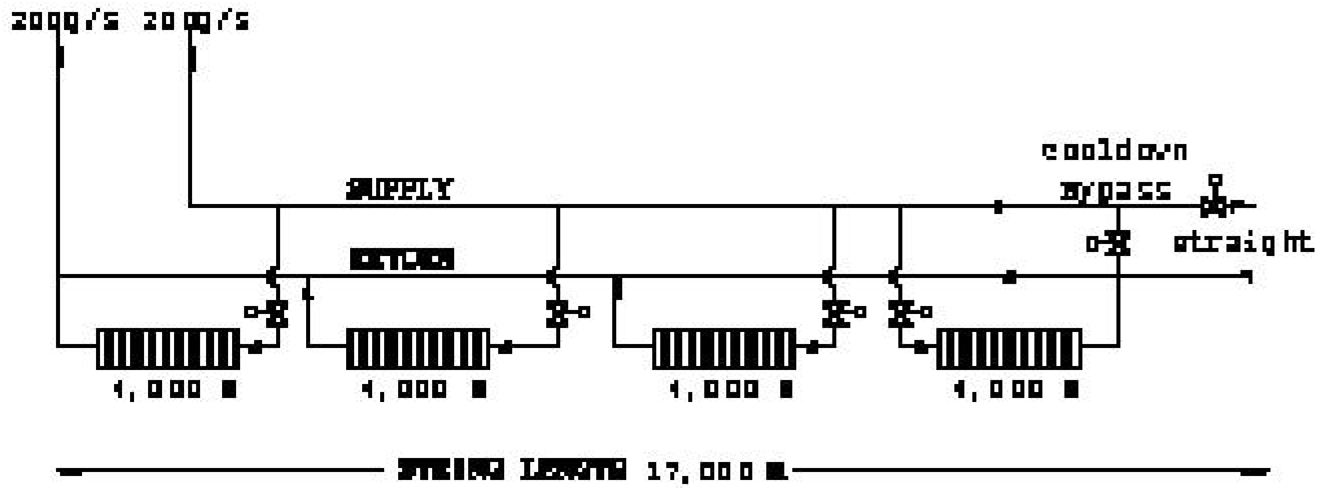

Figure I Layout of a String

This suggests that the SC cost difference between a transmission line designed for 6 K and one for 5K is less that \$30 per meter or \$9000 for a 300 m unit. With the line sizing given in Table I, 40 g/s of flow can be used for heat transfer in the transmission line. With a temperature rise of 4.5-5 K, the sensible heat is about 2 J/g, making the recooler capacity 80 W. With the heat load budget shown in Table IV, 0.26 W/m, one recooler would be required for every 300 m transmission line unit. The cost of this recooler with its J-T valve, level regulation, and instrumentation is surely greater than \$9000. The costs developed for these objects for the SSC was considerably greater than this. Of course the heat load budgeted may be greater than that eventually achieved, and the operating temperature range could be increased, but whether or not it can be argued that the cost for this major simplification of the cryogenic system is actually negative, it is certainly modest and definitely worth it.

This argument only extends the discussion at Snowmass. It is clear that the low-field magnet option to be viable must have a low cost per meter. At the same time, the accelerator will be large. This puts a major pressure on for simplification of all of the accelerator systems. This argues for increased pressures and increased temperatures of operation allowing increased temperature and pressure gradients, increased unit sizes, decreased line sizes and reduced instrumentation requirements. The supercritical system will be cheaper to construct and more robust in its operation than the saturated system, and although attention will be paid in the course of system design studies to the issues raised by the increased operating temperature, the choice now is to adopt the supercritical system as first choice for the 3 TeV Injector. This system will be described in the following sections.

II CRYOGENIC SYSTEM LAYOUT AND OPERATION

For the 3 TeV Injector which is proposed with the modest circumference, at least on an RLHC scale, of 34,000 m, the minimum number of 1 refrigeration station is required. The choice of the number of stations is the balance between the cost of the stations, which have many economies of scale, and the costs of dealing with longer strings. These include the costs of larger piping, cost of underground real estate and installation, the logistics of underground work, the larger heat load and cryogen inventory associated with a unit length of the larger piping, and string voltage and quench protection issues. This is not just an economic calculation, because the costs of surface stations are likely to be political as well as economic. The work at Snowmass showed that piping and electrical string length do not begin to present problems at 20,000 m. In addition, the total refrigeration plant capacity required for the 3 TeV machine is within the current experience. In this case, therefore, the choice is a simple one, and the system can be laid out in two strings connected to a single service station containing both refrigeration plant, power supply, and dump. Table II presents some of parameters of the system.

Table II
System Parameters

Circumference	m	34,000	Number of Sections/String		4
Number of Refrigeration Stations		1	Straight Length	m	1,000
Number of Strings		2	Section Length	m	4,000
Number of Straights		2	Cell length	m	300

The layout of each string is likewise simple. The string consists of a cryogenic pipeline containing supply and return headers. In the current picture of the 3 TeV system, the vacuum jacket of this pipeline doubles as a support member for the magnet. The magnet transmission line is connected in four parallel passes between the supply and the return, and the flow of supercritical helium through each pass is controlled by a valve and flowmeter. The general arrangement is illustrated in Figure I.

Also shown connected in parallel with the four sections of the magnet transmission line is a fifth circuit for one of the two long straights that are part of the layout. This can, of course, go anywhere along the string. Further, it is assumed that the cryogenic pipeline carries across the straight, so that the arrangement can be adapted to suit the final arrangement of all of the components.

Table III
String Heat Load

Load by Type	Number/String	Supply Line W/unit	Supply Line W	Magnet Line W/unit	Magnet Line W	Return Line W/unit	Return Line W
MLI	17,000 m			0.08	1,360	0.32	5,440
Gas Cond.	17,000 m	.002	34	Inc in MLI		Inc in MLI	
Support	17,000 m	.002	34	0.18	3,060	0.10	1,700
Connections	60	1.2	72	5	300	22	1,320
Vac Barrier		2x0.1		2x1		2x6	
Splice		1		1		0	
Connection				2		10	
Control Devices	10			8	80		
Refr. Connection Piping			100		0		400
Total Load per String W			240		4,800		8,860
Total Load for Two Strings W			480		9,600		17,720

A heat load breakdown for one of these strings is given in Table III. This is intended to represent a heat load that can be achieved with straightforward design at low risk. This is thus a high point with which to begin optimization. It is desirable to have a cryogenic system that will operate feasibly at this load so that costs and benefits associated with optimization can be assigned values. The basis on which this load was developed will be discussed below. For the moment consider the total refrigeration requirement in each category and compare it to the process that is outlined in Table IV.

Table IV
Refrigeration and Ideal Power: 400 g/s Total Flow

Point or Load	P bar	T K	h J/g	exergy J/g	Load W	Eff. W/W	Ideal Powr MW	%
Refr. Supply State	4.5	4.548	12.47	1045				
Delivery & Cont. ?P	3.25	4.633	12.47	1107			.0248	1.85
Supply Line Load					812	61	.0496	3.70
Mag. Line inlet	3.25	5.000	14.5	1231				
Magnet Line					9,600	61	.5816	43.4
Mag. Line Outlet	1.90	5.891	38.5	2685				
Return Line Load								
Refr. Return State	1.3	13.60	84.5	4393	18,400	37	.6832	51.0
Total					28,812		1.339	100

The supply stream from the refrigeration plant is 400 g/s at 4.5 bar and 4.55 K. This is divided between the two strings, flowing in the supply line. At 4000 m intervals along this line flows of 45 g/s are drawn off through flow control valves and pass into the magnet transmission line. This is illustrated in Figure I. An initial estimate of the line ?P is 0.4 bar, half of which appears in the first 4000 m, but this calculation is complicated by the bus carried in this line. Table IV shows that a pressure drop of 1.25 bar has been budgeted for the flow in the supply line plus the drop in the control valve. This is adequate to cover the requirements plus a contingency, and as is indicated in the last column of the Table, the ideal power associated with this pressure drop is less than 2% of the total. About an average of 2 J/g (800 W) is budgeted for the heat load of the supply line system and again there is a contingency. This goes into the fluid unevenly, the maximum rise being 3 J/g at the end of the string. Few-percent flow adjustments in the magnet transmission line compensates for this.

In the magnet transmission line, pressure drop is made a virtue since expansion keeps the temperature down. The flow enters at a nominal 3.25 bar and 5 K and exits at 1.9 bar and 5.9 K. The enthalpy increases by 24 J/g, and the density of falls by a factor of 6 along the stream. Computer calculations of the pressure and temperature along the transmission line show that the flow state remains single-phase and keeps well away from the critical everywhere. Calculations also show that the heat transfer coefficients [8] between the conductor and the helium are large enough everywhere that the temperature differences within the transmission line are negligible compared with the end-to-end temperature rise.

This scheme works well over a fairly large range of flow rates. For turn-down of a factor of two, if this is required, a back-pressure valve about 2/5 of the way along the line will have to be added. Calculations so far show that the largest dynamic load in this system is the loss due to conductor splices, and this does not present a significant control problem.

At the exit of the magnet transmission line sections, the flow is collected in the return line. The arrangement shown in Figure I gives a minimum of 65 g/s and a maximum of 155 g/s. This minimizes pressure drop while providing enough flow everywhere to pick up the heat load. The pressure drop has been estimated to be 0.45 bar and 0.6 bar has been budgeted in Table IV. The heat load of this line and shield raises the enthalpy of the flow by an average of 46 J/g. The flow returns to the refrigeration plant at 1.3 bar and 13.6 K.

In addition to the steady operating mode just described, the cryogenic system must be capable of transient modes of operation to accomplish system cool-down, warm-up, and quench recovery. String cool-down in this system is straightforward: After readiness testing, the cooldown process is established by passing helium at 5 bar down the supply line and back through the return line to the refrigeration plant at 1.3 bar. This is a pressure-controlled process, and about 25 g/s will flow when the system is warm All of the section control valves and the bypass valve at the end of the string are open, so there is flow established in all of the circuits. The system has a volume of about 200 cu-m and holds 200 kg of helium. The circulation replaces this inventory every 2-1/2 hours, and purification will be complete in less than a day. After cleanup, the flow from the refrigeration plant is cooled to about 20 K, and cooldown begins. As the cooling wave reaches the outlet of each section of the transmission line, the control valve for that section is closed to keep the cooling wave moving . As the system cools down, the flow rate will increase. The cold mass of the whole system is about 170,000 kg with a heat content of 24 GJ. the first wave cooldown will be complete after 16,500 kg of helium has circulated. This is estimated to take about 4 days. In the third day, therefore, the inlet temperature to the string will be slowly lowered to the operating temperature and the refrigeration process will be supplemented with helium from storage. An important point is reached when the return temperature to the refrigeration plant approaches the 13.6 K operating point. When the flow rate into the string reaches 200 g/s, the pressure will be reduced to the 4.5 bar nominal operating pressure. The operating inventory of the string is about 18,500 kg, and at 200 g/s unbalanced flow, filling will take about 1 day. We can estimate, therefore that something like 2-1/2 days will be required to complete the second wave of cooldown. Adding a day to condition the string to the right temperature profiles, the cooldown will require 7-8 days. This is about the same time that is required by the Tevatron system.

Warming up is the reverse process to cool-down. The first step in warm-up is storage of the inventory. This is done by adjusting the refrigeration plant to supply to the string a flow of 25 g/s at 4.5 bar and 13.6 K, and at the same time expanding the 200 g/s at 4.5 K into the storage tank. This is the same volume flow as the 200 g/s at 4.5 K, and so the inventory comes out of the string at the 200 g/s rate and is re-liquefied by the refrigerator into storage. After the inventory is recovered, the string is warmed up by circulation of gas. Because of the variation of heat capacity of solids with temperature, warm-up waves spread out in a system, and the warm-up process is not as efficient as the cooldown. It is reasonable to assume that a warm-up will take something like two weeks.

The last transient process to be considered here is quench recovery. Quench in the transmission-line magnet is a non-event from the cryogenic point of view because almost all of the magnet stored energy is dumped in the dump resistors. The quench propagation velocity in the conductor is adjusted to be just large enough to provide resistance growth for quench detection. Thus all there is to quench recovery is getting some warm helium out of the transmission line section. The rate at which this can be done is determined by the transit time of the helium through the 4000 m path. This is about two hours. The warm gas in the return line goes directly to the refrigeration plant without disturbing neighboring sections and the refrigeration process has no difficulty in dealing with the small amount of heat involved.

III MAGNET TRANSMISSION LINE DESIGN ISSUES

From a cryogenic point of view, the most important system component is the magnet transmission line. In the system that proposed here the transmission line heat load determines the helium flow required and thus sets the sizes of all of the piping in the system. It has been mentioned above that the heat loads for the cryogenic lines in Table III are suggested as starting points for development. We will here present arguments for these numbers.

The space provided in the magnet iron for the transmission line cryostat is 3 inches diameter, and the superconducting bus has a cross-sectional area of about 3.5 sq-cm. This insulated bus is enclosed in a cryogen-carrying pipe 1.5 inches outside with a wall 0.049 inches, and the inside clear diameter for helium flow is 1.125 inches diameter

The conductor inside the magnet is at a point of unstable equilibrium in the vertical direction, and feels a force gradient of about C = 2 MN/sq-m or 290 lb. per inch of conductor per inch of displacement from the center of the magnet iron. If the cold pipe is supported at intervals by spiders, the field gradient gives rise to an instability in which the current-carrying pipe bends into a sinuous curve with a period twice the support distance. Surprisingly, this problem is one that can be solved exactly. The elastic curve of the pipe is actually a sinusoid, and the unstable support interval, L_o, is given by the root of the first equation when the frequency of the transverse sinusoidal motion of the pipe is zero:

$$-e \cdot w^2 = C - T \cdot \left(\frac{p}{L}\right)^2 - E \cdot J \cdot \left(\frac{p}{L}\right)^4 = 0$$

$$L_0 \cong p \cdot \left(\frac{EJ}{C}\right)^{1/4} \left(1 + \frac{T}{2\sqrt{CEJ}} + \frac{1}{2}\left(\frac{T}{2\sqrt{CEJ}}\right)^2\right)^{1/2} \quad for \frac{T}{2\sqrt{CEJ}} << 1$$

An approximate solution is given in the second equation. Here T is the tension, E is the modulus of elasticity and J is the moment of area of the section of the pipe.

Some of the mechanical parameters of the conductor and pipe are given in Table V. Here the spring constant of the support spiders has been chosen to be a factor of 4 greater than the value at which the pipe is unstable at the support point. This is C·L = 1MN/m. The conductor will not be exactly on the magnetic center of the iron, so there will be an off-center force. This misalignment has been chosen to be at a maximum 0.5 mm, and the force is given in the table. The maximum force that will be felt at the supports is the sum of the off-center force and force due to the weight of the pipe and conductor. This total is about 700 N.

Note that the frequency of the support loaded by the pipe mass is lower than the pipe frequency. Thus the compliance of the support is mass-like rather than stiffness-like at this frequency, and it is certainly necessary to carefully review what is meant by a stiff support. The simple model presented here will have to have some correction terms.

Table V
Magnet Transmission Line Mechanical Parameters

Vacuum Jacket	SS 3 inch od x 0.083 wall	Magnetic Spring Constant, C	$2 MN/m^2$
Inner line	Invar, 1.5 inch od x 0.049 wall	Support stiffness, SC = n·C·L	4 MN/m
Support Interval, L	0.50 m	Max. center misalignment, d	0.5 mm
Max Support Interval, L_o	0.67 m	Max.Off-center force, n·C·L·d/(n-1)	667 N
Conductor area	3.5 sq-cm	Mass of pipe + conductor, ε	4 kg/m
Weight of conductor		2.84 kg/m	Lowest
pipe freq., ω/2p	318 Hz		
Support stiffness factor, n	4	Vert. support freq., $((n-1)\cdot C/\varepsilon)^{1/2}/2p$	195 Hz

To get a feeling for the scope of the support problem, neglect bending and elastic failure, and consider the limiting case of pure tension or compression. The heat flow, Q, can be is related to the stress, σ, and l, the length of the thermal and mechanical path in the simple way shown below.

$$Q = l \cdot \frac{F}{s \cdot l} \quad and \quad SC = \frac{F \cdot E}{s \cdot l} \quad \Rightarrow \quad Q = l \cdot \frac{SC}{E}$$

In these equations λ is the thermal conductivity integral of the support material at 300 K. Clearly, the spring constant varies inversely with the allowed stress and the length, and there are two possible limits to the Q that can be achieved. One is represented in the last equation with the required spring constant, and the second in the first equation with the limiting stress. Thus there is a heat vs allowed stress trade-off for heat loads above the minimum set by what is required to achieve the needed spring constant.

At this point it is useful to look at some properties of real materials, and a likely candidates are fiber reinforced composites. There is a great deal of experience with these materials in the support of cryogenic tankage, in spacecraft, and other similar applications. There is a good understanding of how to design and manufacture these materials and what their fatigue properties are [3,4]. In the application under discussion here fatigue is important , since the 3 TeV injector will cycle on the order of 10^7 times in a 20-year lifetime. Table VI shows some properties of two of these materials [3] and the heat loads that are predicted from them. For both materials the properties are those of uniaxial straps in tension. Small diameter glass-epoxy tubular struts in compression have a somewhat higher cyclic compressive stress limit and carbon somewhat lower.

Table VI
Support Materials

	S-Glass Epoxy	Low-Modulus (LM) Carbon Epoxy
Young's Modulus, E	60 GPa	140 GPa
Thermal Conductivity Integral, λ	150 W/m	550 W/m
Limiting Stress for 10^7 Cycles, σ	0.2 GPa	1 GPa
$Q = \lambda \cdot SC(\text{required})/E$	20 mW/m	31 mW/m
$Q = \lambda \cdot F/\sigma \cdot l$ (l = 0.025 m)	21 mW/m	31 mW/m

It is remarkable that both the stress limited and the stiffness limited heat leaks of both of these materials coincide at the scale of .025 m support length.

A design program for this support must deal with many requirements besides the few fundamental ones considered above. Care must be taken with the way that the loads are transmitted through the support: the way that the magnetic load on the conductor is transmitted to the pipe; from the pipe to the support; from the support to the vacuum jacket; and from the vacuum jacket to the magnet iron. All of these things are important in achieving the needed stiffness. In addition to the magnetic forces the support system must resist assembly and transportation forces and forces due to thermal contraction of the inner pipe during cooldown and warmup, and it must be manufacturable and reasonably cheap (there are 70,000 in the system). The heat leak that is budgeted in Table III, 180 mW/m, is a reasonable place to begin this design process. This provides adequate room for optimization in the heat leak-stress-stiffness space with a good prospect of getting a fully satisfactory result.

Although the support heat load is the most critical in the magnet transmission line, the thermal radiation heat load must also be controlled by the use of multi-layer insulation (MLI). There is little to add on this topic to the discussion at Snowmass. A dimpled mylar MLI system has been used in a number of large systems and is commercially available [1]. It is reported as performing at the level of 0.3 W/sq-m for 40 layers in a 1 cm thickness. This would produce a radiation heat load of 0.06 W/m in the transmission line. In addition, a variable-density MLI system with fiberglass paper spacers has demonstrated an average heat flux of 0.21 W/sq-m in testing on a hydrogen calorimeter with a 310 K warm boundary [2]. There are about 54 layers in this package which is 57 mm thick. The behavior of both of these package is almost that of ideal floating shields. This demonstrates that a reasonably simple understanding of the functioning of these MLI packages is adequate to predict their performance. The radial space available in the transmission line is 17 mm, so the limit as to what can be achieved in the line is something like 0.2 W/sq-m; but the insulation package is interrupted by the supports, providing paths for radiation leakage. Because of this 0.08 W/m has been budgeted for radiation heat load. It is likely that in order to achieve this the supports will have to be integrated with the MIL package at a few levels by means of aluminized mylar cuffs. Again in this case cost optimization will determine the final form that the insulation package will take.

In the discussion so far it has been ignored that it is only in one plane that the conductor feels the decentering gradient. The gradient in the other plane produces centering forces, and the support system can be designed to take advantage of this. However, if this is done means will have to be found to orient the support inside the vacuum jacket and the vacuum jacket within the iron. This is not impossible, clearly. There can be ears or pins in the wall of the outer pipe that engage both the support spiders and the iron. But this has implications for manufacturability, assembly and maintenance as well as performance and cost of the system, and consideration of the cost-benefit optimization of a non-round transmission line system must come after the issues of the round design are more fully understood.

To finish up the discussion of the magnet transmission line, some ideas concerning construction should be mentioned. In the current picture, the accelerator is constructed in 300 m lengths, transported into the tunnel and installed. The 300 m sections of the cryogenic lines are terminated with vacuum barriers and permanently evacuated. This vacuum is maintained during operation by adsorbent and getter packages distributed in the vacuum space. Assembly in the tunnel involves what is termed in the cryogenics business a field joint. The conductor is spliced by soft soldering on a copper mandrel. This also serves to anchor the conductor longitudinally, and longitudinal forces on the inner line are transmitted through the vacuum barrier to the outer jacket. The inner line is connected by welding in place a sliding sleeve. MLI is then installed and a second sleeve is welded to make up the vacuum insulation. This leaves a small volume of vacuum insulation to be roughed out and checked and pumped by sorption material in a frangible capsule. The vacuum barrier in this picture is a piece of stainless steel pipe about 2 inches in diameter with 0.035 inch wall 16 inches long. This has a heat leak of 1 watt that appears in the budget of Table III.

Table VII
Line Sizing and Inventory

Length	**OD**	**Wall**	**Inlet State**			**Outlet State**			**Flow**	**Wght**	**Invntory**
			T	P	Flow	T	P	Flow	Area		LHe
m	in	in	K	bar	g/s	K	bar	g/s	sq-cm	kg/m	cu-m
Magnet transmission line: In 4 parallel 4,000 m flow paths plus straight per string. 34,000 m total length											
4,000	1.5	0.049	5.000	3.25	45.0	5.891	1.90	45.0	6.41	3.99	21.8
Supply line: In 2 strings. Line is 2-1/2 Sch 5 SS pipe carrying conductor. 34,000 m total length											
17,000	2.875	0.083	4.548	4.5	200.0	5.000	4.00	**	33.6	6.13	114.2
Return line: In 2 strings. Line is 3-1/2 Sch 5 SS pipe. 34,000 m total length											
17,000	4.000	0.083	5.891	1.90	**	13.6	1.30	155.0	74.5	4.61	16.9
Total Inventory											152.9
Times 1.2											183.4
											About 50,000 gal.

IV CRYOGENIC SUPPLY AND RETURN PIPING

Here there are two thermally isolated pipelines in a single vacuum jacket. The supply line contains the transmission line return bus and carries the supply flow of cryogen from the refrigeration station. The return line carries the return flow of cryogen back to the refrigeration station. In contact with this pipe is a thermal shield surrounding the supply line. Thus almost all of the heat leak into the pair of lines is taken as sensible heat in the flow of the return line.

These lines will fit into a 12 inch vacuum jacket, so the circumference of the cold boundary is somewhat less than 1 meter. If the same dimpled insulation system is used here, one could expect a radiative heat load of about 0.32 W/m. There is more room for the supports in this cryostat than in the magnet transmission line, and although the weight to be supported is greater, the magnetic forces are smaller. The support problem is therefore much less complicated. The load estimate in Table III is based on having S-glass epoxy support posts for the return line, 7/16 inch in diameter and 4 inches long, a pair every three meters. This provides more than adequate support and scope for optimization. The supply line is supported from the return line by stainless steel straps, 0.02 inches thick, 0.5 inches wide and 3 inches long, a pair every two meters. These straps are pictured as being hinged at top and bottom with pins so that some longitudinal motion is allowed. This is necessary to provide for thermal motions of the lines during cooldown or warmup of the system. Table VII gives the sizes and weights of the cryogenic lines and the helium inventory of the system.

V. REFRIGERATION PLANT

The refrigeration requirements for the 3 TeV machine are given in Table IV. The system operates with a flow of 400 g/s of helium supplied at 4.5 bar and 4.55 K and returning at 1.3 bar and 13.6 K. The total refrigeration load is 28.812 kW and the refrigeration ideal power is 1.339 MW. This ideal power is equivalent to 20.4 kW at 4.5 K, and so the overall capacity is close to the plants that were planned for the SSC and close also the plants to be used for the LHC [5]. There is a further similarity to the LHC in that part of the load is absorbed by sensible heat. In LHC the cold compressor load and part of the load in the beam screen circuit are taken at temperatures above the saturated level. Such a “cold compressor cycle” operates at Thomas Jefferson Laboratory and there has been considerable recent study of this kind of process in the context of TESLA [6] and APT. Thus the refrigeration plant required here lies along the main path of development of large-scale helium refrigeration today, and the both process design and plant engineering issues will be widely familiar ones as a plant for the 3 TeV machine is planned and procured.

To get an idea of what a plant suitable for FNAL might look like, the process for a satellite refrigerator for the 3 TeV application has been investigated [7]. This is a large version of the satellite refrigerator familiar Fermilab with two engines rather than one at the low temperature end to cover the refrigeration required up to the 13.6 K temperature. In order to supply 400 g/ s of flow under the conditions mentioned above, this process requires 103 g/s of liquefier flow at 3 bar and an enthalpy of 14 J/g. This is typical of the Fermilab CHL which will produce approximately 200 g/s of flow with these conditions from the upgraded cold box. The satellite heat exchanger requires 1.03 kg/s of flow at 19 bar and returns the 1.13 kg/s at 1 bar. The engines have inlet temperatures of 15.03 K and 9.82 K and produce work of 11.8 and 4.9 kW respectively at an efficiency of 0.7. Taking a compressor efficiency of 0.5 for the satellite flow, and proportioning the compressor and nitrogen plant power requirement of the CHL, the total power requirement of the process is 6.7 MW for an overall efficiency of 20% of Carnot.

This exercise is not to be taken seriously as a design, but it does provide a set of parameters for a plant that although not optimized, is entirely realizable and provides for all of the operating modes required for the 3 TeV machine. The goal in continuing this work is the development of concepts and simple cost models for a stand-alone and for a satellite plant. These concepts need to include such site-specific requirements as tunnel depth and location relative to the currently installed Fermilab cryogenic plant and utilities.

VI. CONCLUSION

The cryogenic system outlined here is meant to be both an investigation of feasibility and a starting point from which to develop an optimized conceptual design. Of course, optimization proceeds throughout the design of any of the systems of the 3 TeV machine. However, we are not yet to square one in this process. The basic ideas of the Pipetron need enough development to form a beginning that is complete and self-consistent at the most basic technical level. To move in this direction three undertakings present themselves. These are briefly described below:

1) There is a complete enough set of requirements to begin a preliminary design of the magnet transmission line. The goal here is to produce a first-order consistent design together with a complete mechanical analysis and a thermal model. Follow-on to this work is development of a cost model for the transmission line and the design of some appropriate verification and demonstration testing.

2) The supply and return pipeline is simpler in its mechanical requirements that the transmission line. In this case MLI performance is very important, and it is very desirable to develop an appropriate system. The most promising candidate is the dimpled mylar system that has been previously investigated here at FNAL. We should procure supplies and undertake calorimeter testing to confirm previous results and develop competence to handle the system. Then it would be reasonable to build a return pipeline prototype for a demonstration heat leak test. This MLI would also be used in the transmission line.

3) As mentioned above, steady-state calculations of the thermodynamic state of the helium coolant in the transmission line have been carried out. This work should be extended to a dynamic simulation of the thermo-hydraulics of the line. This is needed for two reasons: the first is to support the design of a control system for the helium flow. It is important to confirm that a temperature measurement every 1000 m, which is the system currently contemplated, is sufficient to permit efficient control of the cryogenics under conditions of ramping and other transient operations. The second need is for the further understanding of the stability margin of the conductor and the development of a quench. The Pipetron conductor is to operate in a regime of cooling that is different from that investigated before [9] and unfamiliar in accelerator magnets. This is not discussed above, and it is worth mentioning the issues here.

The Pipetron concept employs a conductor cooled by an internal flow, but one that is to be protected by active detection of a quench and switched ramp-down of the magnet system. This certainly seems to be the most appropriate and economical system that can be employed. The conductor has been designed estimating quench velocity and resistance growth in the adiabatic limit [11], but there is a large amount of helium heat capacity per unit length. The magnetic stored energy is 66 J/g in the helium, and if this is absorbed at constant volume, the final state is at about 28 K (and 9 MPa). The steady state heat transfer in the conductor produces only a small reduction in the adiabatic quench velocity [12], but the heat transfer is large enough so that the flow of the helium accelerates very rapidly after a quench. The heat transfer increases on time scales smaller than that required to produce a detectable quench voltage, and will certainly affect the resistance growth time. This is the same situation that produces stability in cable-in-conduit conductors with static or slowly-flowing helium, but the heat transfer coefficients are clearly larger in that case.

Thus further work is needed to get the design basis of the Pipetron conductor into satisfactory shape. Some quantitative understanding of the minimum quench energy and margin, minimum propagating zone sizes, quench growth resistance, and so on is absolutely needed in order to support a conductor testing program; and if history is a guide, complete control of these issues will be needed to get to a fully finished design.

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